Renewable and Sustainable Energy Reviews 16 (2012) 4927–4937

Contents lists available at SciVerse ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

A review on two-phase ejector as an expansion device in vapor compression refrigeration cycle

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ARTICLE INFO

Article history:
Received 17 February 2012
Received in revised form
11 April 2012
Accepted 18 April 2012
Available online 27 June 2012

Keywords:
Two-phase ejector
Vapor compression
Expansion device
Cooling capacity
COP improvement

ABSTRACT

This paper presents a comprehensive review of two-phase ejector as an expansion device in vapor compression refrigeration cycle over the past two decades. It also covers research opportunities that are still open in the field of two-phase ejectors as an expansion valve. The studies of the application of ejector as an expansion device are relatively scarce compared to the application of ejector as heat-driven refrigeration system. A better understanding of two-phase flow in the ejector is necessary to optimize energy saving of the system. This paper also presents effects of geometric parameters of the ejector as an expansion valve on the performance of vapor compression refrigeration cycle. In addition, the effect of working fluid on the two-phase expansion refrigeration system is covered. The authors predict that the challenge of future research on design of two-phase ejector is how to generate a pressure rise in diffuser for minimum compressor work and optimum COP improvement.

Contents

1. Introduction

There are several ways of improving the performance of vapor compression refrigeration cycle (VCRC) [1]. An improvement in the performance of refrigeration cycle will result in energy saving of the system. The use of the heat exchanger for subcooling and superheating is a conventional method. In the recent time, several researchers apply inverter and control method to regulate the motor rotation of compressor according to cooling load in the cooled-compartment [2–5]. Using an ejector as expansion device is also one of the alternative ways of improving refrigeration cycle performance. This paper presents a review of an ejector used as an expansion device in vapor compression refrigeration cycle.

Typical vapor compression refrigeration cycle uses capillary tube, thermostatic expansion valve and other throttling devices to reduce refrigerant pressure from condenser to evaporator. Theoretically, the pressure drop is considered isenthalpic process (constant enthalpy). Isenthalpic process causes a decrease in the evaporator cooling capacity because of energy loss in the throttling process [6]. To recover this energy loss, isentropic (constant entropy) is required in the expansion process. An ejector can be used to generate isentropic condition in the throttling process.

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http://dx.doi.org/10.1016/j.rser.2012.04.058

Because the phase of the working fluid in the diffuser is a two-phase, an ejector as an expansion device is usually named as two-phase ejector in most papers [6–12], while others refer to it as an ejector-expansion refrigeration cycle (EERC) [6,13,14].

Ejectors have been used in a different applications [15,16]. For example, in the refrigeration systems, ejector is applied to ejector refrigeration cycle (ERC) that is driven by heat which replace compressor in the VCRC. The ERC that was introduced by Le Blanc and Parsons in 1901 [17] have successfully produced a refrigeration effect using an ejector, powered by heat energy. The steam heat is a source of energy, and it is referred to as steam-jet refrigeration system. Sokolov and Hershgal [18] proposed an ERC powered by combination of mechanical and waste heat energy. Economically, their system can be more advantageous over the VCRC if the cost of waste heat is relatively cheap.

The idea of two-phase ejector as an expansion device is not new. In 1931, Gay [19] patented two-phase ejector to minimize throttling losses by replacing conventional expansion valve with ejector. In 1966, the modifications of the patent were proposed by Kemper et al. [20], by using a pump and a heater to increase pressure and temperature of the liquid stream before introducing it to motive nozzle. Newton had improved the idea in the form of other patents in 1972 [21,22], by introducing hot gas from compressor discharge on the liquid stream before entering into motive nozzle. However, numerical analysis on newly emerging EERC in 1990, was conducted by Kornhauser [6].

Research studies on ejector as an expansion device is on the increase since 1990s to date. Numerical analysis and experimental results on EERC showed increasing coefficient of performance (COP) on the VCRC. The increase in COP is caused by reducing compressor work and increasing cooling capacity in the evaporator. Investigation on effect of geometric parameters of ejector, i.e. dimension of throat nozzle, mixing chamber, constant area and diffuser angle is ongoing [10,13,14].

In summary, there are at least three categories of utilization of ejector in refrigeration system. The first is as a substitute for the compressor on the heat-driven refrigeration system, [17,18,23–39], especially with the increasing depletion of fossil fuel reserves. The second is the application of ejectors in the condenser on the vapor compression refrigeration cycle. The idea of this application is still relatively new. Bergander [40,41] claimed that the use of ejectors in the condenser of the vapor compression refrigeration system, which he proposed, was the first. Chen et al. [42] proposed an innovative use of ejector on the vapor compression heat pump cycle for water heating application to improve the heating load at low ambient temperature condition. The position of the ejector in the refrigeration cycle is similar to Bergander's concept of placing between a compressor and condenser. The third is the application of ejectors as a substitute for conventional expansion device in refrigeration systems with the purpose of reducing throttling loss process. The application of ejectors as an expansion device will be discussed extensively in this paper.

The schematic diagrams of the three applications of an ejector on the vapor compression cycle are shown in Fig. 1. On the heat-driven refrigeration, in Fig. 1(a), the primary flow (high pressure), secondary flow (low pressure) and the diffuser are points 1, 6 and 3, respectively. Also, the ejector on the condenser, in Fig. 1(b), the primary flow, secondary flow and the diffuser are points 2, 7 and 4, respectively. In addition, in an ejector as an expansion device, in Fig. 1(c), the primary flow, secondary flow and the diffuser are points 3, 8 and 5, respectively

Fig. 2 illustrates the working fluid phase of three application of an ejector in Ph (pressure-enthalpy) diagram. It shows that working fluid of the primary and the secondary flows are in different working phase for each application. The working principle of an ejector is similar to a pump. A pump needs external power to take fluid to a higher pressure. An ejector uses a high pressure fluid (primary flow) to induce fluid from low pressure (secondary flow) to a higher pressure at diffuser outlet.

Table 1 depicts the phase and temperature range estimation of the working fluid through the ejector in the outlet of nozzle, suction and diffuser. Also the table shows the uses of ejector in the various applications on the refrigeration system.

2. Operation principles of ejector as an expansion device

Elbel and Hrnjak [43] was the first person who used an ejector to pump liquid water to the reservoir of steam engine boiler. Keenan and Neumann [44] developed a numerical method using one-dimensional continuity, momentum and energy equations to investigate the ejector performance. In their analysis, it was

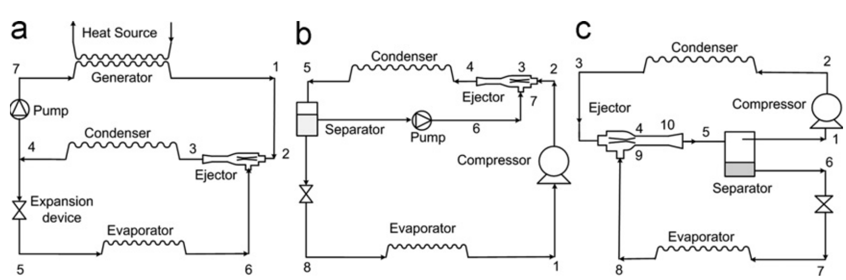

Fig. 1. Schematic diagram of three applications of an ejector on the vapor compression refrigeration cycle. (a) Ejector on the heat-driven refrigeration, (b) ejector on the condenser and (c) ejector as an expansion valve.

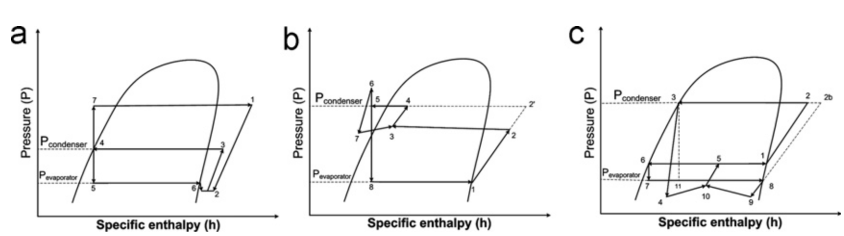

Fig. 2. Ph diagram of three applications of an ejector on the vapor compression refrigeration cycle. (a) Ph diagram of ejector on the heat-driven refrigeration, (b) Ph diagram of ejector on the condenser and (c) Ph diagram of ejector as an expansion device.

Table 1
Phase and temperature range of the working fluid in the ejector.

Application on the refrigeration cycle	Nozzle flow	Suction flow	Diffuser flow	Objectives
Heat-driven	**Vapor**	**Vapor**	**Vapor**	To replace the compressor on the vapor compression refrigeration cycle.
Temperature range (°C) (Estimation)				
– Only for air conditioning	> 80	0–15	35–55	
On the condenser	**Vapor**	**Liquid**	**Two-phase**	To improve the COP by reducing the compressor work.
Temperature range (°C) (Estimation)				
– For air conditioning	< 35	< 20	35–55	
– For freezer	< 35	< 20	35–55	
Expansion device	**Liquid**	**Vapor**	**Two-phase**	To improve the COP by lowering the compressor work and increasing the cooling capacity.
Temperature range (°C) (Estimation)				
– For air conditioning	< 5	0–15	> 0, < 15	
– For freezer	< (−5)	(−40)–(−5)	> (−40), < (−5)	

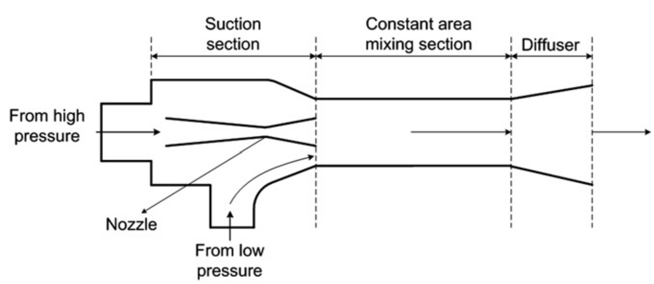

Fig. 3. Constant-area mixing ejector.

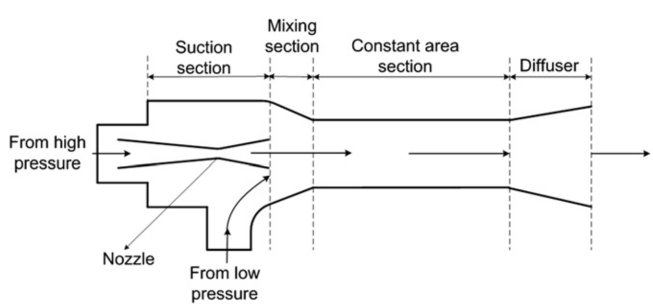

Fig. 4. Constant-pressure mixing ejector.

difficult to analytically explain the mixing phenomena. Therefore, their study was continued by Keenan et al. [45], who concluded that there two types of ejector, namely constant-area mixing ejector and constant-pressure mixing ejector.

Fig. 3 shows a constant-area mixing ejector that has three sections: a nozzle, a constant area mixing, and a diffuser. The mixing between primary flow from high pressure and secondary flow from low pressure occur at inlet constant area. Also, Fig. 4 illustrates a constant-pressure mixing ejector with four sections: a nozzle, a mixing, a constant area and a diffuser 4. The mixing between primary and secondary flow occur at mixing section or suction chamber. Development of mathematical modeling on flow inside the ejectors is presented by He et al. [46]. Their paper described the evolution process and the current status of the mathematical models on ejectors.

In the quantitative investigation research, the constant-pressure mixing ejector has a better performance than that of the constant area [34,45]. As a result, a constant-pressure mixing ejector is generally used in various refrigeration applications, especially in ejector refrigeration systems [27,30,33,35,37]. However, Yapici and Ersoy [29] found that for the same operating temperature, the constant-area mixing ejector has higher COP than that the constant-pressure mixing ejector. Therefore, in the last decade, most researchers use constant-area mixing ejector for the numerical and experimental studies on EERC in their research [10–14].

Capillary tube, thermostatic expansion valve and automatic expansion valve are expansion devices that are commonly used in refrigeration system [47–49]. These expansion devices are essentially a restriction offering resistance to flow so that the pressures drop, resulting in a throttling process. Basically there are two types of expansion devices, namely constant-restriction and variable-restriction [48]. Capillary tube is constant-restriction type of expansion device having a small diameter. Thermostatic expansion valve and automatic expansion valve are variable-restriction type, in which the opening of valve depends on the cooling load in the evaporator. The capillary tube is the simplest and cheapest of the expansion device. It is used in small capacity refrigeration system. For medium to large cooling capacity, thermostatic or automatic expansion valve is use as an expansion device. The use of capillary tube and other conventional expansion devices causes the loss of kinetic energy which arises due to pressure drop. Theoretically, this throttling is assumed as an isenthalpic process. Utilization of ejector as an expansion device will change the isenthalpic to isentropic process as shown in Fig. 5. Isenthalpic process is from point 3 to 11, while isentropic process is from point 3 to 4.

Fig. 5 shows the EERC and standard cycle on Ph diagram. In the figure, refrigerant flow on the Ph diagram of standard cycle is point 8, 2b, 3, 11 and 8. While, on the EERC there are two flow, primary and secondary flow. The primary flow is circulated by a compressor through condenser, ejector and separator (point 1, 2, 3, 4, 10, 5 and 1), whereas the secondary flow circulates in the expansion valve, evaporator, ejector and separator (point 6, 7, 8, 9, 10, 5 and 6). The primary and secondary flow mix at constant area and diffuser (point 10 and 5). As shown in Fig. 5, the pressure at point 1 is higher than that of suction pressure in the standard cycle (point 8). This means that the compressor work of the ejector expansion cycle is lower than that of the standard cycle.

The operation principle of two-phase ejector as an expansion device is similar to the ejector function in other applications, in which the primary flow from high pressure induces the secondary flow from low pressure in the suction section of ejector and brings it to a higher pressure at diffuser. Fig. 6 depicts the refrigerant flow, pressure and velocity profile inside an ejector. The motive flow or primary flow from high pressure is accelerated and expands through motive nozzle, from 3 to 4. Very high speed flow causes pressure drop at point 4 and induces fluid from secondary flow at point 8 to 9. The two flows mix at inlet of mixing section and become one stream in the constant area. The mixing stream flows through point 10 to the diffuser. In the diffuser, the refrigerant experiences deceleration as a result of pressure increase (point 5).

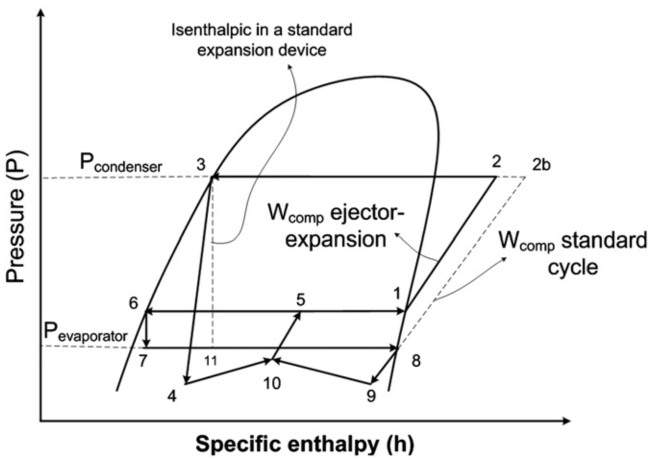

Fig. 5. Ph diagram of EERC and standard cycle.

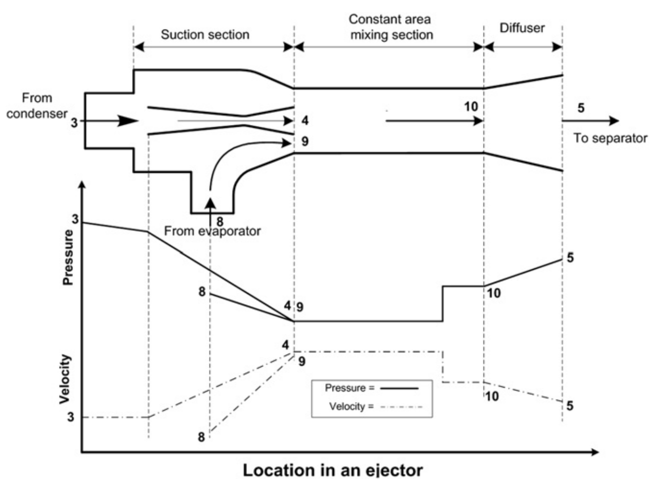

Fig. 6. Refrigerant flow, pressure and velocity profile inside an ejector.

Fig. 7 shows the schematic diagram of the standard cycle and an EERC. The refrigerant at point 5 flows to separator. The vapor refrigerant in the separator circulates through compressor (point 1 to 2) as a primary flow, whereas the liquid refrigerant from separator circulates through expansion valve and evaporator (point 6, 7 and 8), as a secondary flow.

The performance of the VCRC can be explained by Ph diagram shown in Fig. 5. In the standard refrigeration cycle, the COP can be calculated as

$$COP_{std} = \frac{Q_e}{W_{comp}} = \frac{\dot{m}_{evap}(h_8 - h_{11})}{\dot{m}_{comp}(h_{2b} - h_8)} \tag{1}$$

Due to $\dot{m}_{evap} = \dot{m}_{comp}$, Eq. (1) becomes

$$COP_{std} = \frac{(h_8 - h_7)}{(h_{2b} - h_8)} \tag{2}$$

This condition does not occur in the ejector as expansion device, because the mass flow rate of evaporator is not the same with the mass flow rate of compressor. Also, from Fig. 5, the COP of ejector expansion device is

$$COP_{ejc} = \frac{Q_e}{W_{comp}} = \frac{\dot{m}_{evap}}{\dot{m}_{comp}} \frac{(h_8 - h_7)}{(h_2 - h_1)} \tag{3}$$

or

$$COP_{ejc} = \omega \frac{(h_8 - h_7)}{(h_2 - h_1)} \tag{4}$$

where, ω is entrainment ratio of an ejector.

Eq. (4) shows that an increase in COP is determined by the entrainment ratio of ejector. While the ejector entrainment ratio is influenced by the design of ejector, thus a better understanding of ejector design is crucial in improving the COP system.

In the application of the EERC, the ejector performances are characterized by using several parameters, such as entrainment ratio (ω) and pressure lifting ratio (P_{lift}) and ejector efficiency (η_{ejec}). From Eqs. (3) and (4), the entrainment ratio (ω) is

$$\omega = \frac{\dot{m}_{evap}}{\dot{m}_{cond}} \tag{5}$$

Fig. 8 is used to determine pressure lifting ratio (P_{lift}) and ejector efficiency (η_{ejec}). The figure is derived from Fig. 5.

$$P_{lift} = \frac{P_{diff,out}}{P_{evap,out}} = \frac{P_1}{P_8} \tag{6}$$

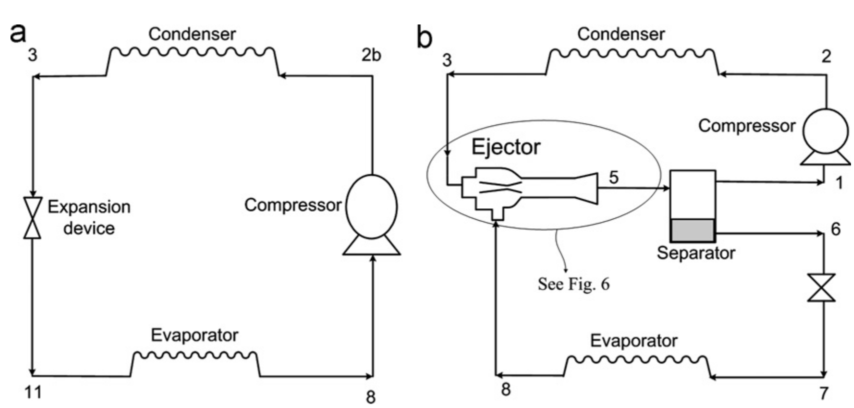

Fig. 7. Schematic diagram of vapor compression refrigeration cycle: (a) standard cycle and (b) ejector-expansion cycle.

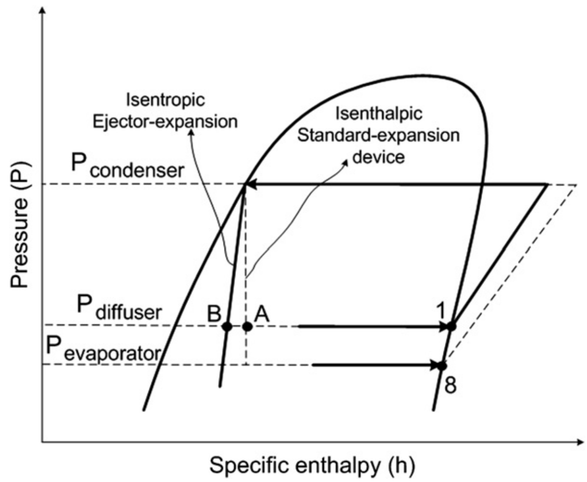

Fig. 8. Expansion and compression of driving and driven flow inside two-phase ejector.

$$\eta_{ejc} = \omega \cdot \frac{(h_1 - h_8)}{(h_A - h_B)} \tag{7}$$

Point A and B in Fig. 8 are enthalpy at diffuser pressure when process isenthalpic and isentropic, respectively. Meanwhile, point 1 and 8 are the compressor suction and outlet evaporator pressure, respectively.

Entrainment ratio (ω), pressure lifting ratio (P_{lift}) and ejector efficiency (η_{ejec}) are useful parameter for measuring the performance of an ejector. These quantities should be as large as possible for improvement in the efficiency of the system [11]. High ejector pressure lifting ratio decreases the compression ratio of the compressor. Increasing the mass entrainment ratio will reduce compressor mass flow rate for a given cooling capacity. Ejector efficiency increases when mass entrainment ratio and/or pressure lifting ratio increase. However, the entrainment ratio cannot be increased as high as possible, because it will cause the flow of refrigerant on primary flow to reduce. A high entrainment ratio will cause the primary flow as a driven-flow to be weak.

3. Thermodynamics modeling

Before Kornhauser [6] proposed thermodynamic numerical approach on EERC, the working principle of ejector has been studied by researchers, however this principle was not fully understood. Most studies was performed in that time used one-dimensional model and three approaches were applied for the analysis of the ejectors, namely mixing at constant-pressure, mixing at constant-area and combination of constant-pressure and constant-area mixing. Using a constant-pressure ejector, Kornhauser gave the following assumptions:

i. The mixing in the suction chamber occurs in the constant pressure and below evaporator pressure.
ii. Properties and velocities are constant over cross section (one-dimensional).
iii. The refrigerant condition is in thermodynamic quasi-equilibrium.
iv. Deviation processes from adiabatic reversible process can be expressed by efficiencies.
v. Pressure drop in piping, evaporator, and condenser, also kinetic energy outside ejector is negligible.
vi. No heat-transfer, except in the evaporator and condenser.
vii. Refrigerant leaving evaporator and condenser is saturated vapor and liquid, respectively.

Most researchers follow the assumptions contained in the list above.

3.1. Governing equations

Kornhauser [6] applied the three general governing equations to analyze the nozzle, mixing and diffuser section. The three governing equations are as follows:

$$\text{Conservation of mass}: \sum \rho_i V_i A_i = \sum \rho_o V_o A_o \tag{8}$$

$$\text{Conservation of momentum}: P_i A_i + \sum \dot{m}_i V_i = P_o A_o + \sum \dot{m}_o V_o \tag{9}$$

$$\text{Conservation of energy}: \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} \right) = \sum \dot{m}_o \left(h_o + \frac{V_o^2}{2} \right) \tag{10}$$

where, "i " is input and "o" is output.

Kornhauser [6] also used three efficiencies for each ejector section, namely the efficiency of the nozzle (η_n), suction (η_c) and diffuser (η_d). The definition of the three efficiency of each part of the ejector based on Fig. 5 is as follows:

$$\eta_n = \frac{(h_4 - h_3)}{(h_{4,is} - h_3)} \tag{11}$$

$$\eta_s = \frac{(h_9 - h_8)}{(h_{9,is} - h_8)} \tag{12}$$

$$\eta_d = \frac{(h_{5,is} - h_{10})}{(h_5 - h_{10})} \tag{13}$$

where, "h" is enthalpy, subscript "is" is isentropic process and numbers indicate points in Fig. 5.

These quantities (η_n, η_s, η_d) should be as large as possible for more efficient ejector. The ideal ejector has efficiency value of one (1) and this means that the process in the ejector is isentropic. Based on the numerical analysis, Kornhauser [6] found that efficiency of the diffuser (η_d) has the highest influence on the COP improvement, while efficiency of the nozzle (η_n) has the lowest effect on the COP improvement over standard cycle.

3.2. Calculation algorithm

Kornhauser [6] proposed a numerical procedure to calculate COP improvement on EERC over the standard cycle. Using Fig. 5 and efficiencies definition in Eqs. (11)–(13), the flowchart of his procedure is shown in Fig. 9. The iteration process is as follow:

i. Select the pressure in the outlet of constant area ejector (P_{10}). The pressure at P_{10} must be lower than the evaporator pressure.
ii. Calculate the enthalpy and velocity of refrigerant at outlet motive nozzle (h_4 and V_4) and at suction nozzle (h_9 and V_9) using efficiency definition and energy conservation equations.
iii. Guess flow ratio (r), mass flow rate ratio between mass flow rate in the compressor ($\dot{m}_{comp}$) to the total flow ($\dot{m}_{evap} + \dot{m}_{comp}$).
iv. The enthalpy and velocity at the outlet of constant area ejector (h_{10} and V_{10}) is calculated using equations of energy and momentum conservation, respectively.
v. Entropy in the outlet of constant area ejector (S_{10}) is determined from h_{10} and P_9.
vi. The enthalpy at the diffuser outlet (h_5) is calculated using equations of efficiency definition and energy conservation.

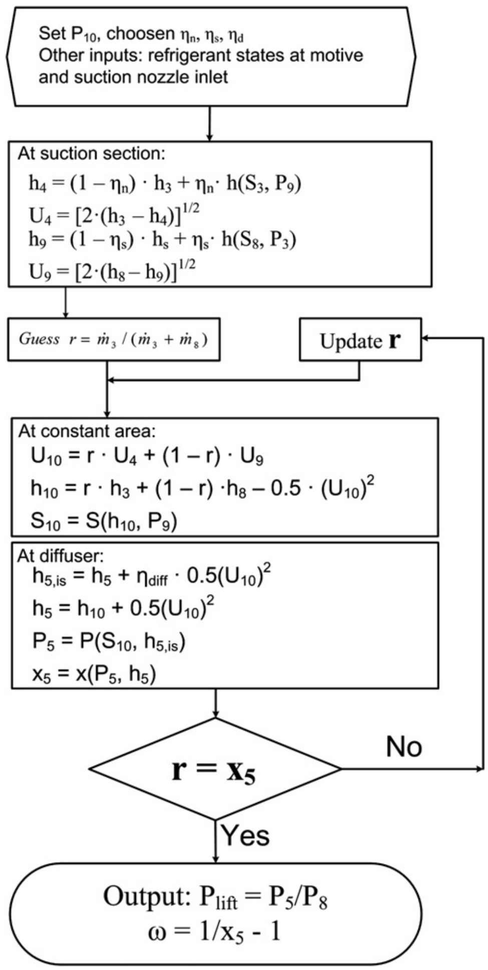

Fig. 9. Flow chart of performance calculation of the EERC according to Kornhauser [6].

vii. The pressure at the diffuser outlet (P_5) is determined from S_{10} and h_5 (at isentropic process $h_{5,is}$).
viii. The vapor quality of refrigerant at the diffuser outlet (x_5) is determined from P_5 and h_5.
ix. The iteration is repeated till flow ratio (r) equal to the vapor quality of refrigerant at the outlet diffuser (x_5).

Nehdi et al. [10] manipulated the algorithm (in Fig. 9) in order to investigate the effect of geometric parameter on the performance of EERC. The manipulation which they performed to acquire the area ratio ($\Phi = A_{10}/A_4$) of the ejector and its influence on the system. Their manipulations begin from the division of the flow in the ejector to the evaporator and the condenser, and obtained the following equation:

$$\dot{m}_{cond} + \dot{m}_{evap} = \rho_{10} V_{10} A_{10} \tag{14}$$

where

$$\dot{m}_{cond} = \rho_4 V_4 A_4 \tag{15}$$

$$\dot{m}_{evap} = \rho_9 V_9 A_9 \tag{16}$$

where, ρ is the density of working fluid; V the velocity of working fluid; A the cross-sectional area

Furthermore, using the conservation of momentum equation they obtained Eq. (17):

$$(P_{10} - P_4)A_{10} = \dot{m}_{cond} V_4 - (\dot{m}_{cond} + \dot{m}_{evap}) V_{10} \tag{17}$$

Table 2
Critical properties of refrigerants.

Refrigerant	Chemical formula	Category	Designation	T_{cri} (°C)	P_{cri} (MPa)
Difluoro-dichloro methane	CF_2Cl_2	CFC	R12	112.04	4.11
Difluoro-monochloro methane	CHF_2Cl	HCFC	R22	96.12	4.99
Tetrafluoro ethane	CF_3CH_2F	HFC	R134a	101.15	4.06
Propane	C_3H_8	HC	R290	96.60	4.25
Isobutane	$(CH_3)_3CH$	HC	R600a	134.70	3.64
Ammonia	NH_3	Natural	R717	132.25	11.33
Carbon dioxide	CO_2	Natural	R744	31.06	7.38

Combining the above Eqs. (14)–(17), they obtained Eq. (18)

$$\frac{P_{10} - P_4}{0.5\rho_4 V_4} = 2\left(\frac{1}{\Phi}\right) - 2(1+\omega)^2 \left(\frac{\rho_4}{\rho_{10}}\right)\left(\frac{1}{\Phi}\right)^2 \tag{18}$$

By manipulating Eqs. (14)–(18) and using equations at suction section and at diffuser in Fig. 9, the effect of area ratio on the COP improvement can be calculated.

Sarkar [14] also manipulated Karnhouser's numerical analysis to investigate effect of area ratio (ϕ) ejector on the EERC using natural refrigerant, such as ammonia, propane and isobutene. His definition of the area ratio (ϕ) is slightly different from Nehdi et al. [10], which is

$$\phi = \frac{A_4 + A_9}{A_{10}} \tag{19}$$

To obtain an optimum area ratio, Sarkar [14,50] modified Eqs. (8)–(10), and yielded Eqs. (20)–(22) in the constant-area mixing section:

$$\rho_4 A_4 V_4 - \rho_9 A_9 V_9 = \rho_{10}(A_4 + A_9)V_{10} = 1 \tag{20}$$

$$P_9(A_4 + A_9) + \frac{1}{1+\omega} V_4 + \frac{1}{1+\omega} V_9 = P_{10}(A_4 + A_9) + V_{10} \tag{21}$$

$$\frac{1}{1+\omega}\left(h_4 + \frac{V_4^2}{2}\right) + \frac{1}{1+\omega}\left(h_9 + \frac{V_9^2}{2}\right) = h_{10} + \frac{V_{10}^2}{2} \tag{22}$$

By manipulating Eqs. (20)–(22) and equations at suction section and at diffuser in Fig. 9, the effect of area ratio (ϕ) on the COP improvement can be obtained.

4. Improvement of COP on the standard cycle

The unique nature of the ejectors on the other areas encouraged the experts to apply the ejectors in the field of refrigeration [17–22]. The improvement COP EERC over standard cycle is defined as

$$COP_i = \frac{COP_{ej} - COP_{std}}{COP_{std}} \tag{23}$$

where, COP_{ej}=COP ejector and COP_{std}=COP standard

Based on the working pressure on the condenser, standard vapor-compression refrigeration cycle can be classified into 2 groups. The first is conventional cycle, namely the condensation temperature below the critical temperature of the working fluid. The second is transcritical cycle, where the condensation temperature is above the critical temperature of the working fluid. The working fluid of conventional cycle is either natural or synthetic refrigerants. Natural refrigerants that are commonly used are ammonia (R717), propane (R290) and isobutane (R600a) [14], and synthetic refrigerants are R12, R22, R410A and R134a [6,7,10,13], while transcritical working fluid is carbon dioxide

(CO_2) [11,50–52]. Due to the negative impact of synthetic refrigerant on the environment, the study of CO_2 as an alternative refrigerant are revived. In fact, CO_2 has been in use since late 19th century but its existence is displaced by the family of synthetic fluorocarbon refrigerants. And finally, CO_2 (R744) as a refrigerant, is abandoned in the decade of 1950 [51,53].

The critical properties of several refrigerants are shown in Table 2. This Table indicates that critical temperature of CO_2 (R744) is 31.06 °C, below condensing temperature of the standard cycle of vapor compression refrigeration cycle, usually 40 °C or above.

Fig. 10 shows Ph diagram of the conventional and transcritical standard vapor compression refrigeration cycle.

4.1. Improvement on the conventional cycle

Since its introduction in the 1930s, refrigerant from the family of fluorocarbons dominate the use of the working fluid in vapor compression refrigeration cycle. At that time the fluorocarbons considered harmless to humans and safe for the environment. Refrigerant CFCs (chlorofluorocarbons), HCFCs (hydro chlorofluorocarbons) and HFCs (hydro fluorocarbons) are synthetic working fluid. Propane, isobutene and ammonia are natural working fluid.

In 1990, Kornhouser [6] was the first researcher who performed numerical analysis on the EERC to investigate the performance improvement on VCRC. There are seven synthetic refrigerants, namely CFCs: R11, R113, R114, R500 and R502; HCFCs; R22 and one natural R717 (ammonia) which he studied. From numerical analysis, he was found that R502 had the highest COP improvement than any other refrigerant. The COP improvement in the R12 was approximately 21% over standard cycle. It was also found that the COP improvement decreases when the evaporator temperature increases. In other words, an increment in the COP is higher in freezers compared to the air conditioners. Calculation method which he developed has been followed by most researchers in the field of ejector-expansion cycle [10,13,14,50].

Four years later, Kornhauser and Menegay [54] received a patent on how to increase velocity of flow at the motive nozzle in order to improve performance of the system. When the liquid refrigerant flows through the motive nozzle, part of it will change to vapor. Since the density of the vapor is much lower than the liquid, so the volume of the gas will reduce the amount of liquid in the nozzle. This condition causes a decrease in flow rate of refrigerant in the motive nozzle. The vapor entering the motive nozzle is in form of large bubbles. A breaker bubbles device is small diameter tube installed before converging–diverging nozzle. When large bubbles pass through a small diameter, it breaks up into smaller bubbles. Thereafter, the smaller bubbles enter the converging–diverging nozzle. The diameter of a breaker bubbles is larger than that of the motive nozzle throat but much more smaller than that of liquid line on the condenser [55].

Due to depletion of ozone layer and global warming effect of CFCs refrigerant on the environment, Harrell and Kornhauser [56] performed experiment of ejector as an expansion device in the system that use R134a (HFCs) as refrigerant. The experiment reported that the COP improvement over standard cycle ranges from 3.9% to 7.6%. This result is lower than the numerical analysis. Using ideal ejector, it is able to achieve 23% over standard cycle. The cause of different result due to the phenomena of single-phase, which is used as a numerical analysis, is different from two-phase flow through the ejector. Based on this result, the two-phase phenomena in the ejector must be studied in order to achieve better performance of using ejector as an expansion device.

In an attempt to make improvements on the performance of two-phase ejector, Menegay and Kornhouser [55] performed an experiment using a bubbly maker tube installed upstream of the nozzle to reduce the non-equilibrium thermodynamic losses in the ejector nozzle. This concept is the same with their patent [54]. By using an air conditioner with R12 (CFCs) as refrigerant with cooling capacity of 3.5 kW, the COP improvement using bubbly flow tube is 3.2–3.8%, and only 2.3%–3.1% without bubbly maker. This result is not as good as expected, and they recommend more investigation in this area.

Disawas and Wongwises [7] carried out the experiment to investigate the effect of heat source and heat sink temperature on the performance of refrigeration cycle using ejector as an expansion device. The experiment used a constant-pressure mixing chamber and convergent–divergent motive nozzle, with nozzle throat diameter of 0.9 mm. On testing with R134a, the heat source temperature was varied from 6 to 18 °C, with the increment of 2 °C, while the heat sink temperature was varied from 25 to 40 °C, with the increment of 5 °C. They found that the COP improvement is lower when the heat sink temperature is increased. Also, the compressor pressure ratio and discharge temperature is lower than that of the standard cycle.

Wongwises and Disawas [8] continued their research above [7] focusing more on the effect of condenser temperature on the COP improvement of EERC. They used water to cool the condenser in their experiment. The inlet cold water temperature on the condenser was varied from 24 to 40 °C, with the increment of 2 °C. They showed the experimental data graphically that the COPs of standard cycle and EERC decrease when the inlet cold water temperatures increase. They also found that COP of the EERC was higher than that of standard cycle, especially at low inlet cold water temperatures.

Chaiwongsa and Wongwises [9] continued their research by testing it with similar equipment. Different from their previous researches [7–8], they vary the throat diameter of the nozzle: 0.8, 0.9 and 1.0 mm. They found that the throat diameter of the nozzle of 0.8 mm yielded the highest COP, and the lowest COP was achieved when nozzle throat diameter was 1.0 mm. They also found that motive nozzle with throat diameter 1.0 mm produced a higher primary flow rate and recirculation ratio than other nozzles.

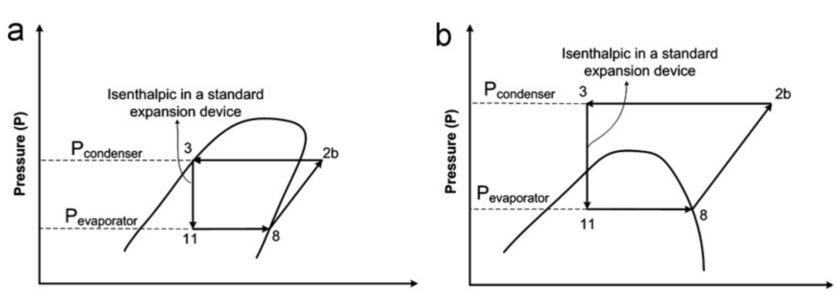

Fig. 10. Ph diagram of vapor compression refrigeration cycle standard: (a) conventional cycle and (b) transcritical cycle.

Nehdi et al. [10] presented numerical analysis of geometric ejector effects on the performance of the system using twenty synthetic refrigerants. They introduced a geometric area ratio (Φ), and ratio of mixing chamber to primary nozzle throat area. Also, they claimed that their analysis on the effect of geometric parameter of ejector is the first. It was concluded that the maximum COP is obtained when the optimum Φ is around 10. For Φ optimum, refrigerant R141b achieved the highest COP. The COP improvement over the standard cycle is 22%.

Bilir and Ersoy [13] performed a computational analysis on the performance improvement of ejector expansion cycle over standard cycle. Computational procedures they have adopted is easy to be followed by beginners who are interested in exploring ejector as an expansion device. Computational methods that they did in fact is similar to that of Kornhauser [6]. Using an R134a refrigerant, the COP improvement of the expansion cycle over standard cycle is 10.1–22.34%. They also found that the COP improvement increases when the condenser temperature increases. This means that the use of ejector instead of an expansion valve is more advantageous in the air-cooled condensers than that of water-cooled condensers.

Sarkar [14] carried out a numerical analysis of area ratio (ϕ) effect on the COP improvement. Three natural refrigerants, namely ammonia, propane and isobutane, were used in his computational. He also investigated the influence of three parameters of ejectors: the entrainment ratio, pressure lift ratio and geometry area ratio on the performance of ejector-expansion cycle. He found that the area ratio and entrainment ratio optimum differ from each refrigerant. The optimum area ratio is maximum for ammonia, however minimum for isobutane. The optimum entrainment ratio is maximum for ammonia, but minimum for propane. The area ratio optimum (ϕ_{opt}) for R600a, R290 and R417 are 6.23, 3.36 and 11.02, respectively. The highest COP enhancement produced by isobutane was 21.6%, 17.9% and 11.9% for isobutane, propane and ammonia, respectively.

Ersoy and Bilir [57] investigated exergy analysis on the EERC. They found that the exergy destruction of each component of the EERC always lower than that of standard cycle. The total exergy destruction rate of standard cycle is 39.613 kJ/kg, while on EERC was 16.345 kJ/kg, in other word there is reduction in exergy destruction by 58.7%. They also found that the optimum area ratio ejector increases when the efficiencies of ejector components decreases. The highest efficiency of ejector resulted in an optimum area ratio. Each ejector design has an optimum area ratio, and there is only one optimum area ratio for a given evaporator and condenser temperature on the EERC.

Table 3 shows the summary of the researches on EERC in the vapor compression refrigeration cycle using conventional refrigerants.

4.2. Improvement on the transcritical cycle

Due to negative effect of the synthetic refrigerant to environment, CO_2 refrigerant is revive as an environmental friendly working fluid. Also, losses due to expansion in vapor compression refrigeration cycle using CO_2 is relatively high compared to the cycle using conventional refrigerants [11]. As an illustration in the vapor compression with conventional refrigerants, typically, the expansion ranges from 2 to 0.2 MPa, depending on the working fluid and evaporating temperature. On the other hand, for transcritical, the expansion ranges from 10 to 2 MPa. In other words, theoretically, recovery potential of expansion losses caused by throttling process on transcrtical system is higher than that of conventional one.

Liu et al. [58] presented a numerical analysis using Kornhauser's iteration method on ejector-expansion refrigeration with CO_2 as a refrigerant. An ejector is used as an expansion valve in transcritical cycle to recover the kinetic energy losses in the throttling process. They found that the suction pressure of compressor is higher than that of standard cycle, resulting in lower ratio of compression and therefore reducing compressor work on the ejector expansion cycle. The COP improvement on this transcrtical cycle ranges from 6 to 14% over standard cycle.

Ozaki et al. [59] carried out an experiment on the automotive air conditioning using R744 as a refrigerant to improve COP system. The experiment yielded COP improvement 20% over standard cycle. They also showed that two-phase ejector has an advantage of control at high-side pressure compared to transcrtical expander.

Li and Groll [60] performed a thermodynamic numerical analysis using a constant-pressure missing model ejector. The effect of the entrainment ratio and the pressure drop in suction chamber were investigated. For given conditions, the ejector-expansion cycle has a 7–18% improvement over the standard

Table 3
The summary of papers of ejector as an expansion device in the vapor compression refrigeration cycle using conventional refrigerants.

Authors	Working fluid	COP improvement	Remarks
Kornhauser (1990) [6]	R11, R12, R22, R113, R114, R500, R502, R717	21%	Numerical analysis
Harrell and Kornhauser (1995) [55]	R134a	3.9–7.6%	Experimental on the air conditioning system.
Menegay & Karnhouser (1996) [54]	R12	3.2–3.8%	Experimental on the air conditioning system.
Disawas and Wongwises (2004) [7]	R143a		Experimental on the freezer and air conditioning system. The improvement of COP becomes relatively less as the heat sink temperature increases.
Wongwises and Disawas (2005) [8]	R143a		Continuing the previous experimental, focusing more on the effect of condenser temperature on the COP improvement.
Chaiwongsa and Wongwises (2007) [9]	R134a		D_{nt}=0.8 mm yields the highest COP.
Nehdi et al. (2007) [10]	±20 refrigerants	22%	R141b is the highest COP. Area ratio (Φ) optimum is about 10.
Bilir and Ersoy (2009) [13]	R134a	10.1–22.34%	Numerical. The COP improvement increases when the condenser temperature increases.
Sarkar (2010) [14]	R717, R290, R600a	R600a=21.6% R290=17.9% R417=11.9%	Numerical analysis. ϕ_{opt} (R600a)=6.23 ϕ_{opt} (R290)=3.36 ϕ_{opt} (R417)=11.02

cycle. Decreasing the entrainment ratio causes increase in the suction pressure, this condition reduces pressure ratio of the compressor, and further increases the COP improvement. It is also obtained that the COP improvement increases when the pressure drop in the suction chamber increases. The pressure drop in the suction chamber is determined by geometry of the ejector and operating temperature condition. Using the transcritical cycle of CO_2 as refrigerant, it was found that for typical air conditioning operation, the improvement of the system over the standard cycle could be more than 16%.

Deng et al. [52] performed energy and exergy analysis of the EERC on transcritical cycle. On the energy analysis, they found that the improvement of COP of the EERC achieved 22% over standard cycle. The exergy loss on each of components on the EERC and standard cycle for high-side pressure of 8.7 MPa is shown in Fig. 11. This figure shows that the exergy loss of throttling process in the standard cycle is 34.29% of the total exergy loss. The exergy loss of expansion in the EERC consists of throttling and ejection losses of 1.36% and 28.38%, respectively. The sum of the two losses is 29.7% of the total system exergy loss, and is lower than that of standard cycle by 34.29%. There is also a reduction in exergy loss in the compressor of EERC over standard cycle, from 25.69% to 23.73% of the total exergy loss.

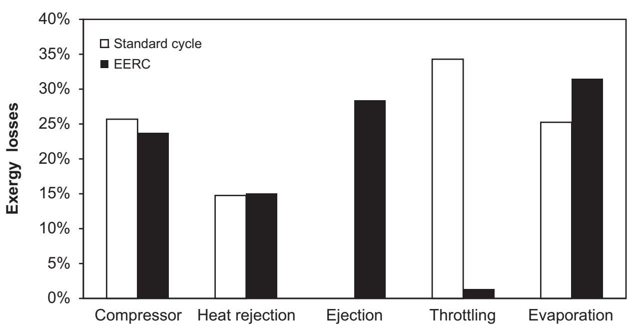

Fig. 11. Exergy losses in the standard cycle and EERC for a high-side pressure of 8.7 MPa.

Yari and Sirousazar [61] performed a theoretical analysis on ejector-expansion transcritical CO_2 two-stage refrigeration cycle. They proposed a new system using an internal heat exchanger and an intercooler to improve the performance. Their simulation results showed that the COP and the second law efficiency of the new cycle increased by about 55.5% and 26%, respectively, with the operating conditions of 10 °C evaporator temperature and 40 °C gas cooler outlet temperature. They also found that the total exergy destruction rate of the new ejector-expansion cycle (two-stage) was 4% lower than that of standard cycle.

Elbel and Hrnjak [11] were the first researchers that introduced variable two-phase ejector by installing a needle in motive nozzle to regulate diameter of throat nozzle. The needle allows the regulation of the high-side pressure. The results showed that the cooling capacity and COP of the EERC could be maximized by regulating the needle opening in the nozzle throat. The improvement of cooling capacity and COP was up to 8% and 7%, respectively. It was found that the maximum COP was achieved when the area of the nozzle throat is reduced. This result was similar with the experiment carried out by Chaiwongsa and Wongwises [9]. In their experiment, they found that the COP of EERC was the best when the throat diameter is 0.8 mm, compared to 0.9 and 1.0 mm. Besides, introducing variable-area nozzle throat, Elbel and Hrnjak investigate the effect of diffuser angle on the EERC. Three different diffuser angles, 5°, 10° and 15° were used in their experiment. The smallest angle (5°) yielded the highest ejector efficiencies.

Elbel [12] presented a review of the ejector application on the refrigeration system and experimental results of the EERC on the CO_2 refrigeration. He used a number of diffuser modules similar to his previous study [11], and used four different lengths of the mixing chamber (constant-area): 7.5, 33.5, 57.5 and 82.5 mm. He found that the length of the constant-area significantly affects the ejector efficiency. The shortest constant-area yielded efficiency up to 15% and the efficiency drop to 7% for longest one. The secondary flow and pressure lifts increase for shorter

Table 4
The summary of papers of ejector as an expansion device in the vapor compression refrigeration cycle using CO_2 refrigerant.

Authors	Working fluid	COP improvement	Remarks
Liu et al. (2002) [49]	R744 (CO_2)	6–14%	Numerical.
Ozaki et al. (2004) [57]	R744 (CO_2)	20%	Experimental on the automotive air conditioning.
Li and Groll (2005) [50]	R744 (CO_2)	16%	Numerical on the air conditioning system.
Deng et al. (2007) [58]	R744 (CO_2)	22%	Numerical.
Yari & Sirousazar (2008) [59]	R744 (CO_2)	55.5%	Numerical analysis on two-stage refrigeration cycle using internal heat exchanger and an intercooler.
Elbel & Hrnjak (2008) [11]	R744 (CO_2)	7%	Experimental on the air conditioning system. The best result is for 5 °C angle of diffuser.
Elbel (2011) [12]	R744 (CO_2)	7%	Experimental: the diffuser angle and the mixing section length have significant effect on the COP improvement.

Table 5
The optimum geometric of an ejector based on the numerical analysis and experimental that yields the highest improvement COP.

Authors	**Type of study**	**Working fluid**	**Dimension of ejector parameter**
Chaiwongsa and Wongwises (2007) [9]	Experimental	R134a	D_{nt}=0.8 mm (Diameter of nozzle throat)
Nehdi et al. (2007) [10]	Numerical	R141b	$\Phi \approx 10$
Elbel and Hrnjk (2007) [11]	Experimental	R744	Diffuser angle is 5 °C
Sarkar (2010) [14]	Numerical	R600a	ϕ_{opt} (R600a)=6.23
		R290	ϕ_{opt} (R290)=3.36
		R417	ϕ_{opt} (R417)=11.02
Elbel (2011) [12]	Experimental	CO_2	Lconstan-area=7.5 mm

constant-area. The improvement of cooling capacity and COP over standard cycle was 7% and 8% respectively, the same with his previous findings.

Table 4 shows the summary of the researches on EERC in the vapor compression refrigeration cycle using CO_2 refrigerants.

4.3. Optimum results

Based on all these numerical and experiment analysis it can be concluded that the geometric parameter of ejector yielded the best improvement of COP system, as shown in Table 5.

Despite the fact that optimum geometry was obtained to produce the maximum COP improvement, however there are still significant difference between the results of calculations with the experiment. Therefore, a more intensive research is still needed in order to acquire optimum results and can be practically implemented in the real system.

Thermodynamics analysis by Bilir and Ersoy [13] concluded that the COP improvement of EERC will be optimum for high temperature condenser. Therefore, the use of EERC is appropriate for areas that have relatively high temperature environments, such as tropical countries and desert area.

5. Conclusions

Numerical analysis and experiment results show that using two-phase ejector as an expansion device causes the improvement of COP on the vapor compression refrigeration cycle. Thermodynamic analysis showed that the improvement of COP is achieved above 20%, however, none of the experimental method yielded improvement over 10%. The effect of geometric dimensions of the ejector, such as throat of motive nozzle, suction chamber, constant area and diffuser on the improvement of the system is still an interesting research topic by majority of the researchers. The optimum geometric parameters obtained so far can be used as a foothold for another research for a better improvement. One-dimensional analysis that was developed by Kornhauser also has to be modified, so that the difference between numerical and experimental is relatively small.

The performance of the ejector is influenced by evaporation and condensation temperatures; therefore, future research challenge is on how to design an ejector that can be regulated by the dimensions of the nozzle section and the suction chamber in accordance with the required conditions. This is due to these parts that are relatively critical in determining the efficiency of ejectors. The use of variable two-phase ejector will accelerate the implementation of this device to replace a conventional expansion device. However, to accomplish this, more intensive study on the characteristics of two-phase ejector as an expansion device is still required.

Acknowledgments

The authors wish to acknowledge the support provided by Universiti Teknologi Malaysia: FRGS Fund no. 78686, Ministry of Higher Education (MOHE) Malaysia.

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